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RESEARCH MEMORANDUM

COMPARISON OF OUTSIDE-SURFACE HEAT-TRANSFER COEFFICIENTS FOR CASCADES OF TURBINE BLADES

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

WASHINGTON
July 17, 1950

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COMPARISON OF OUTSIDE-SURFACE HEAT-TRANSFER COEFFICIENTS FOR CASCADES OF TURBINE BLADES

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SUMMARY

Available literature from heat-transfer investigations on cascades of turbine blades was surveyed and the results from each investigation are presented according to the correlation procedure used by each investigator. A comparison of these results was made using the Nusselt equation with the Reynolds number defined by either the inlet velocity and pressure or the average of the velocities and the pressures around the blades.

A correlation of the results obtained from the impulse blades investigated was improved by using the average Reynolds number. By using this correlation procedure, the results from all the turbine-blade investigations with the exception of one blade, which had a high degree of reaction, could be represented by a mean line with a maximum deviation of ±15 percent.

The results from an investigation with a reaction blade were correlated by introducing the temperature ratio with an exponent of approximately 1/3 in the Nusselt equation.

INTRODUCTION

In order to attain high gas-turbine inlet temperatures or operation at current temperatures with turbine blades constructed of nonstrategic materials, the blades must be cooled below temperatures at which extreme losses in strength occur. Evaluation of turbine-blade cooling is dependent on a knowledge of the outside-surface heat-transfer coefficient as well as many other factors.

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The heat-transfer process for flow along flat plates and normal to and inside cylinders has received considerable experimental study and the results have been satisfactorily correlated (for example, reference 1). Application of these results to turbine blades is questionable because a heat-transfer-correlation procedure for flat plates and cylinders is simplified by the similarity in shape parameters and proportionalities in flow parameters.

A limited number of experimental heat-transfer investigations have been made on static cascades of both impulse- and reaction-type turbine blades (references 2 to 6). In all cases, the average outside-surface heat-transfer coefficients were determined and an individual correlation of the results from each cascade was made. The correlation procedures used were, in general, the same as that used for results from similarly shaped bodies but the correlation parameters were defined differently. In most cases, the investigations were made using low temperature differences.

A study of the outside-surface heat-transfer process made at the NACA Lewis laboratory presents a collection of available heat-transfer results obtained from investigations in which outside-surface coefficients were determined for static turbine-blade cascades. The correlation of the results from each experiment, as presented by the individual investigator, is included with a brief description of the experimental conditions and evaluation of the results. In addition, a correlation procedure is proposed and applied to the available results and a comparison is made using this procedure. Factors are pointed out that seemingly influence the correlations. Further experimental research to investigate such factors is therefore indicated.

Acknowledgement is made to the General Electric Company and the British National Gas Turbine Establishment for helpful cooperation.

CORRELATION EQUATION

Experiments with heated or cooled bodies in a gas stream indicate that the lengthy functional relation for heat transfer obtained from dimensional analysis can, in many cases, be reduced to the simple Nusselt equation

$$Nu = C_I (Re)^{C_{II}} (Pr)^{C_{III}} \qquad (1)$$

(All symbols are defined in the appendix.)

This equation has been conventionally used to relate the heat-transfer process for forced convection. The exact definition of the variables in each parameter, however, has not been established. In the available literature pertaining to heat-transfer investigations with turbine blades, the heat-transfer coefficient has been based on the difference between the effective gas temperature and the surface temperature. Various temperatures have been used to define the gas properties and the density. In some recent work (reference 1), better correlation was obtained when both the gas properties and the density were defined by the surface temperature. For most correlation work, the pressure defining the density and the velocity have been taken in the main stream at the inlet to the test body. In some cases, however, the exit or mean velocity and pressure have been used. Moreover, the characteristic length in the Reynolds and Nusselt numbers has, in general, differed for each different geometric shape investigated. One characteristic length, the perimeter divided by π, has been frequently used for correlating results from turbine blades.

Equation (1) has been sufficient for correlating results from heat-transfer investigations with similarly shaped bodies or one particular body when the temperature ratios (ratio of the free-stream temperature to the surface temperature) are near 1. In most cases, the definition of the various parameters therefore seems to have little or no effect on the correlation. Results from various complex shapes, such as turbine blades, with variable temperature ratios indicate that this simplified equation is inadequate for correlation and that more specific definitions of the parameters are needed.

Most heat-transfer investigations with turbine blades have been performed in order to obtain immediate design data for the particular turbine blade investigated. A correlation of the particular data was therefore required but objective correlation with all other available results was not investigated. A correlation of the results from each individual heat-transfer investigation has been made herein using equation (1).

PRESENTATION OF AVAILABLE DATA

Heat-transfer results for static turbine-blade cascades have been obtained by the NACA (reference 2), in Great Britain (reference 3), and in Germany (references 4 and 5). Drawings of all the blades and passage configurations used in these investigations and, in addition, unpublished data from a single heat-transfer investigation by the General Electric Company are presented in figure 1.

In each investigation the results were correlated using the Nusselt equation (equation (1)). The procedure used for defining the parameters in the Nusselt equation and the approximate temperature ratio are given in table I. In addition, table I shows the inlet static gas temperature, the number of blades, and the ratio of the chord to the spacing for each investigation.

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The results of the investigation of reference 2 are shown in figure 2(a). These results have a maximum deviation of approximately ±10 percent from the line represented by the equation on figure 2(a). These data were obtained using a cascade of symmetrical impulse blades that were heated by conduction from an electrically heated dummy-wheel section. Cooled air was passed over the external surface of the blades. The Mach number at the blade outlet was varied from 0.3 to 1.0. A recovery-factor calibration curve for the blade was experimentally determined and the heat-transfer coefficient was computed by assuming a one-dimensional temperature distribution (in a radial direction) in the experimental blade.

The results of the unpublished investigation are presented in figure 2(b). The results can be represented by the Nusselt equation although the parameters that have been plotted differ from those in the Nusselt equation. Only seven data points were obtained; however, the deviation from a straight line is small and the results are well represented by the equation given on figure 2(b). The tests were performed using air at low velocities passed over the cascade of impulse blades, which were heated by steam flowing into the interior of the blades. Because the air velocities were low, the effective gas temperature was approximately equal to the total gas temperature. The heat-flow rate was computed by measuring the quantity of steam condensing inside the blade.

Shown in figures 2(c) and 2(d) are the results of the investigations of reference 3. These investigations were conducted using both heated air and combustion gases passed over the cascade of blades. The blades were cooled by water flowing through an annular passage near the blade surface. The recovery factor was assumed as 0.85 and the heat-transfer coefficient was determined from the heat-flow rate, which could be computed by measuring the temperature rise in the cooling water. Initially the over-all heat-transfer coefficient was computed and the data points plotted, as shown in figures 2(c) and 2(d). The solid line representing these results was then corrected to a curve for which the heat-transfer coefficient was based on the difference between the effective gas temperature and the average blade temperature by computing the blade-to-water coefficient from existing formulas. The results in figure 2(c) were obtained using a cascade of impulse blades and the results in figure 2(d) were obtained using a cascade of reaction blades. The blades for both cascades were the same but the blade stagger changed to increase or to decrease the percentage reaction. The results are represented by the equations given on figures 2(c) and 2(d) with a maximum deviation of approximately ±10 percent.

Figure 2(e) presents the results of experiments published in reference 4. The cascade was so designed that the angle of attack and the ratio of chord to spacing could be varied. The results presented are for two ratios of chord to spacing. The results for the larger ratio are accurately represented by the corresponding equation in figure 2(e). The results from the larger spacing deviate somewhat from the corresponding equation. The investigators apparently established this equation by comparing trends obtained from the various results at different blade spacings and angles of attack. Air was passed through the cascade of blades and the experimental blade was heated by steam flowing through five holes in the blade. Because the air velocities were relatively low, the effective gas temperature was approximately equal to the total gas temperature. The heat-flow rate was determined from the drop in enthalpy of the steam as it passed through the blade.

The results of the investigation of reference 5 are shown in figure 2(f). These results are represented by the equation in figure 2(f). Air was passed through the cascade of impulse blades and the experimental blade was heated by electric heating elements located in the blade. Because the air velocities were relatively low, the effective gas temperature is approximately equal to the total gas temperature. The heat-transfer coefficient was determined from the heat-flow rate, which was computed from the power input to the heating elements.

All results except those from the reaction blade of reference 3 were obtained from investigations at temperature ratios that were approximately constant and near 1. The results from the reaction blade of reference 3 were obtained at temperature ratios from approximately 1.0 to 2.0. For this investigation, a temperature-ratio effect was detected.

The need for a representative procedure of correlating heat-transfer results from turbine blades is evidenced by the variations in the equation that has been used to represent the available experimental results.

COMPARISON OF AVAILABLE DATA

Comparison with Inlet Reynolds Number

The correlation of forced-convection heat-transfer results for gases flowing normal to a heated or cooled body has commonly involved using the Nusselt equation with the Reynolds number defined by the main-stream pressure and velocity at the immediate inlet to the body. In addition, the characteristic dimension has frequently been represented by the hydraulic diameter of an equivalent cylinder. For a turbine blade, this quantity is taken as the perimeter divided by π. This procedure has therefore been used to compare the available data. These results are shown in figure 3; the gas properties and the density have been defined by the average blade temperature. This method of defining the gas properties and the density is used herein because of the results presented in reference 1. In reference 1, a correlation of the results from heat-transfer investigations with air flowing inside tubes was obtained using the Nusselt correlation equation when the gas properties and density were defined by the surface temperature; whereas a separate relation was required for each temperature ratio when other temperatures were used.

For figure 3, the results from the reaction-blade investigations of reference 3 have been reduced to a temperature ratio of 1, which is a ratio comparable to those for the other investigations. The heat-transfer rates for the reaction blades are, in some cases, greater than those for the impulse blades although the slopes (which would be 0.5 for a laminar boundary layer and 0.8 for a turbulent boundary layer) of the curves indicate that the impulse blades have a much larger percentage of turbulent boundary layer, the type that promotes the higher heat-transfer rate. The slopes of the curves representing the results from the four impulse blades are all similar, indicating that the percentage of the surface over which there is a turbulent boundary layer is similar. Nevertheless, the heat-transfer rate for these impulse blades varies as much as 45 percent. It is doubtful that this difference could be attributed to experimental error. Apparently, the correlation procedure is incomplete and must be modified to incorporate variables that influence the comparison obtained in figure 3. A modification of this procedure to obtain a complete correlation cannot be made until more extensive theoretical developments and experimental investigations have been completed. In view of current heat-transfer theory, some improvements can be incorporated and comparisons partly justified. Nevertheless, the correlation procedure of figure 3 is probably the most convenient procedure for design purposes and is adequate for correlating results from investigations on one particular cascade of blades operating at a constant temperature ratio. Figure 3 can be used for predicting heat transfer to a cascade by selecting a curve representing the results from a similar blade and blade arrangement.

An argument against defining the average Nusselt number by the Reynolds number based on the inlet velocity and pressure can be advanced by visualizing two cascades of different blades and passage configurations but with similar boundary layers. With the same blade spacing in the two cascades and inlet conditions and blade temperatures, the local heat-transfer coefficient of a region on one blade in each cascade will differ because the local Reynolds number distribution differs. Because the local heat-transfer coefficient differs, the average Nusselt number will, in general, differ. For these two cases, identical inlet Reynolds numbers then yield different heat-flow rates. Apparently, this Reynolds number is in itself insufficient. In order to establish the relation defining the average heat-transfer coefficient, the local coefficients could be expressed by the equation

$$Nu_x = C_{IV} \, Re_x^{\,C_V} \tag{2}$$

and the integrated average determined. The integration cannot be evaluated at present because it is so difficult as to be impractical to express the velocity and pressure around a turbine blade with the independent variable x. It seems reasonable that the integration would yield an expression involving the Reynolds number defined by the averages of the velocities and the pressures around the blade. The use of this average Reynolds number in the correlation equation may improve and present a more reasonable comparison of various results than that obtained using the inlet Reynolds number.

Comparison with Average Reynolds Number

Determination of average Reynolds number. - The available heat-transfer data from turbine-blade cascades have been reworked so that the results could be compared using the average Reynolds number in the correlation equation. Because the velocity and pressure distributions were not experimentally obtained, they were calculated using theoretical methods and the average velocity and pressure were determined as the integrated mean. The stream-filament theory for a compressible fluid (reference 6) was used to determine the local velocities on the blade profile in the channeled portion of the blades. The velocities in the regions of the leading and trailing edges, over which the stream-filament theory does not apply, were estimated by a circulation check (reference 6).

A sample velocity distribution calculated for a reaction blade is shown in figure 4. The stream-filament theory applies over the surface in the regions between O_1 and O_2 and between O_1' and O_2'. For the remaining portion of the blade, a circulation check was made to fair in the velocity curve. The circulation around the leading edge is determined as the product of the inlet velocity, the spacing, and the sine of the acute angle between the inlet-velocity vector and the cascade line. The calculated circulation must check that represented by the difference between the area under the velocity curve from O_2 to the leading edge and from the leading edge to O_2'. The procedure followed was therefore to fair in the velocity curve in order to fulfill such a check. Similarly, for the trailing edge, the circulation represented by the difference in areas from the trailing edge to O_1 and from O_1' to the trailing edge must check that determined from the product of the outlet velocity, the spacing, and the sine of the acute angle between the outlet-velocity vector and the cascade line.

Nondimensional plots of the theoretical velocity distributions for the various blades investigated are shown in figure 5.

The pressure distribution was determined from the velocity distribution by assuming the total temperature and the total pressure as constant throughout the blade passage.

The velocity and pressure distributions were computed for at least three inlet-state conditions for each blade investigated. The inlet Reynolds number and the corresponding average Reynolds number were then computed for each inlet-state condition. The corresponding value of $Nu_{av}/Pr_{g,b}^{1/3}$ was then determined from figure 3 using the calculated inlet Reynolds number and the line for the particular blade. By this method, at least three values of $Nu_{av}/Pr_{g,b}^{1/3}$ and the corresponding average Reynolds numbers were determined for each blade investigated. Thereby, a plot similar to figure 3 could be established.

Application of average Reynolds number. - The results from the investigations with the impulse blades are compared in figure 6.

The conversion from the inlet Reynolds number to the average Reynolds number is completed in figure 6(b). The maximum variation between the results has been reduced from the original 45 percent in figure 6(a) to 23 percent in figure 6(b).

The results from the investigations of reference 3 for both impulse and reaction blades are compared in figure 7 with the abscissa as the inlet Reynolds number in figure 7(a) and the average Reynolds number in figure 7(b). The boundary layer is apparently laminar for the reaction blade and partly turbulent for the impulse blades, as indicated by the slopes of the respective curves. Although the agreement between the two sets of results in figure 7(a) is better than that in figure 7(b), figure 7(b) is a more logical representation of the results because the predominating laminar boundary layer of the reaction blade is expected to be accompanied by lower heat-transfer coefficients for comparable conditions.

Figure 8 is included to show a comparison of all the results computed using the average Reynolds number as the abscissa. The results from the investigation of reference 4 with two blade spacings have been reduced to a common line by this correlation procedure. The results from a flat plate with both a laminar and a turbulent boundary layer (reference 7) are also shown in figure 8. The characteristic length for the flat plate has been changed from the plate length to the perimeter divided by π where the perimeter is taken as twice the blade length. The results from all of the turbine-blade investigations with the exception of the reaction blade of reference 3 can be represented by a mean line on figure 8 with a maximum deviation of ± 15 percent.

Characteristic length. - Because the local heat-transfer coefficient is defined by equation (2), where the characteristic length is the distance x, it seems reasonable to define the average heat-transfer coefficient by the Nusselt equation, where the characteristic dimension is the total surface length from the leading edge to the trailing edge. By integration of equation (2) to express the average heat-transfer coefficient over a surface using simple prescribed velocity distributions, it can be shown that the characteristic length in the resulting Nusselt equation is the total surface length. For turbine blades where the air flows over two surfaces, an average of the two characteristic lengths could be used. This average is expressed as the perimeter divided by 2. The use of this average length rather than the perimeter divided by π cannot, however, improve the comparison obtained in figure 8 because a conversion from one length to the other is a multiplication of all values of both the ordinate and abscissa by a constant (in effect, a simple axis shift). Improvements in the correlation can only result if a characteristic length that cannot be expressed as the product of a constant and the perimeter is used. Equation (2) could possibly be extended to include the effects of geometry parameters by selecting a characteristic length that is representative of the passage configuration as well as the blade shape.

Additional Factors Affecting Correlation

Several parameters other than those in the Nusselt equation, which are subsequently discussed, are known to influence heat transfer to turbine blades. The Nusselt equation as used in figure 8 is therefore expected to correlate the results only partly.

Pressure gradient. - A factor, other than those used in the correlation procedure of figure 8, which may influence the heat-transfer rate is one that takes into account the geometric configuration of the body. Such a term is the Euler number m, which is defined as the ratio of the pressure forces to the momentum. The effect of Euler number on heat transfer for a laminar boundary layer over a wedge having a velocity distribution corresponding to $V_x = C_{VI}\, x^m$ is shown in reference 8 and in figure 9. The ordinate represents C_{IV} of equation (2) with C_{IV} as 0.5. Figure 9 shows that by varying the Euler number from 0 to 1, the local Nusselt number for a given local Reynolds number is increased by approximately 70 percent. The average Nusselt number, however, as determined by integrating equation (2) with C_{IV} as the ordinate and C_V as 0.5, for a given average Reynolds number is increased by slightly less than 20 percent. The effect, as shown in figure 9, may not be typical of those effects applicable for turbine blades with velocity distributions quite different from $V_x = C_{VI}\, x^m$; however, it does indicate the order of magnitude of pressure-gradient effect, which might be typical for a surface with a laminar boundary layer. No solutions are available for pressure-gradient effect in a turbulent boundary layer.

It seems possible to explain in part the trends in figure 8 by a pressure-gradient effect. For example, by comparing the results from flat plates with those from turbine blades it is apparent that a turbine blade with the same percentage of laminar boundary layer (slopes corresponding) as a flat plate will have the much higher heat-transfer rate. In general, the differences in figure 8 are larger than is predicted by the results of figure 9. When equation (2), the velocity distribution, and the pressure-gradient effect of figure 9 are applied to the results from the reaction blade of reference 3, the pressure gradient seemingly could account for one-half of the difference between these results and those from a flat plate with a laminar boundary layer. In addition, the correlation curves for the reaction blades investigated are approximately within the ordinate range, which expresses the results from the impulse blades although the impulse blades have the larger percentage of turbulent boundary layer over their surface. This comparison of the heat-transfer rates, also, can be explained by the more favorable pressure gradient on a reaction blade (fig. 5), which increases the heat transferred through the laminar boundary layer.

Temperature ratio. - Another factor that may influence heat transfer is the ratio of the free-stream temperature to the surface temperature. The temperature ratio may influence the heat transfer in two ways:

(1) It may have a direct effect on the temperature distribution in the boundary layer. Theoretical solutions of references 9 and 10 have shown that for a flat plate with a laminar boundary layer this effect is small for temperature ratios of the order of those used for the experiments reported herein.

(2) The temperature ratio may also have the effect of stabilizing (for a cooled body) or destabilizing (for a heated body) the laminar boundary layer resulting in a shift of the transition region.

Experimental heat-transfer investigations with turbine blades at temperatures appreciably different from the gas temperature are limited to those of reference 3 using a cascade of reaction blades. In order to gain knowledge, the data from this investigation have been reworked with the temperature ratio incorporated in the correlation equation as shown in figure 10. The Reynolds number exponent is taken from figure 2(d). The inlet Reynolds number and the perimeter divided by π as the characteristic length are used for convenience. The results can be represented by expressing the ordinate proportional to the temperature ratio to approximately the negative one-third power. This exponent is much larger than that predicted by the theory of references 9 and 10 for a flat plate. Currently no definite explanation for this large exponent is available. In the case of reaction blades, it may be advisable to apply a temperature-ratio correction factor similar to that of figure 10.

Additional factors that may influence heat transfer to cascades of turbine blades are transition, separation, and centrifugal effects in the boundary layer. Transition from a laminar boundary layer to a turbulent boundary layer is accounted for in the correlation procedures that have been used by the exponent of the Reynolds number. For a more complete understanding of results from turbine blades, a more precise method of interpreting transition effects is needed. For better correlations, a method of introducing transition in the correlation must be developed. The effect of transition on the results of figure 8 is, in most cases, to increase the heat-flow rate for an impulse blade above that for a reaction blade. Because little is known concerning heat transfer in the separated regions, investigations should be made to study this problem.

CONCLUDING REMARKS

The use of an average Reynolds number improved the correlation of results from static cascades of impulse turbine blades. This correlation procedure, which appears reasonable from a theoretical basis, accounts in part for the effects of blade shape and flow-passage configuration.

A reaction blade with a laminar boundary layer had a much greater heat-transfer rate than a flat plate with a laminar boundary layer. This difference together with the results presented indicated the existence of pressure-gradient effects, which seemed greater in magnitude than explainable by the simple theory available. The need for both experimental and theoretical studies of the pressure gradient effects was indicated.

The temperature ratio seemed to have a significant effect on heat transfer with reaction blades. The results from an investigation with a reaction blade were correlated by introducing the temperature ratio with an exponent of approximately 1/3 in the Nusselt equation. More extensive studies with various types of blade are needed to investigate this effect.

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In order to determine rotational effects, actual turbines, instead of the static turbine-blade cascades used, should be investigated and the results analyzed with results from static cascades.

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APPENDIX - SYMBOLS

The following symbols are used in the calculations and the figures.

A	flow area, sq ft
a	velocity of sound, ft/sec
c	chord, ft
$C_I, C_{II}, \ldots, C_{VI}$	arbitrary constants
c_p	specific heat at constant pressure, Btu/(lb)(°F)
D	characteristic length, ft
g	acceleration due to gravity, ft/sec^2
H	heat-transfer coefficient, Btu/(hr)(sq ft)(°F)
k	thermal conductivity, Btu/(hr)(ft)(°F)
L	surface length from leading edge to trailing edge, ft
l	perimeter of blade, ft
m	local Euler number, $\frac{x}{\rho_{g,x} V^2_{g,x}} \frac{\partial p}{\partial x}$
Nu	Nusselt number, HD/k_g
Nu_x	local Nusselt number, $H_x x/k_{g,x}$
Nu_{av}	average Nusselt number, $\frac{H_{av} \frac{l}{\pi}}{k_{g,b}}$
p	absolute static pressure, lb/sq ft

$\overline{p}$	defined by $\overline{p} = 1/2\ (p_{g,i} + p_{g,o})$, lb/sq ft
Pr	Prandtl number, $3600\ c_{p,g}\mu_g/k_g$
R	gas constant, ft-lb/(lb)(°F)
Re	Reynolds number, $\rho_g V_g D/\mu_g$
Re_{av}	average Reynolds number, $\dfrac{p_{g,av}V_{g,av}\frac{l}{\pi}}{Rt_b\mu_{g,b}g}$
Re_i	inlet Reynolds number, $\dfrac{p_{g,i}V_{g,i}\frac{l}{\pi}}{Rt_b\mu_{g,b}g}$
Re_x	local Reynolds number, $\rho_{g,x}V_{g,x}x/\mu_{g,x}$
t	static temperature, °R
t_m	temperature defined by $t_m = 1/2\ (t_{g,e} + t_b)$, °R
V	absolute velocity, ft/sec
w	weight flow, lb/sec
x	distance along surface from leading edge, ft
β	angle formed by inlet-velocity vector and line perpendicular to cascade line, deg
μ	absolute viscosity, slugs/(ft)(sec)
ρ	mass density, slugs/cu ft

Subscripts:

av	average
b	blade or at blade temperature
c	coolant

e	effective
f	film
g	gas
i	inlet
o	outlet
t	throat
w	wall or at wall temperature
x	local position corresponding to distance x
1, 2, . . . , 5	references

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TABLE I - SUMMARY OF VARIABLES USED IN HEAT-TRANSFER INVESTIGATIONS AND CORRELATION PROCEDURES

Investigator	NACA	General Electric	British	British	German
Reference	2		3	4	5
Temperature ratio, $\frac{t_{g,i}}{t_w}$	0.9	0.8	1 to 2	0.83	0.8
Evaluation temperature for: Viscosity	Film[1]	Inlet stream	Average blade	Inlet stream	Inlet stream
Density	Inlet stream	Throat[2]	Mean[3]	Inlet stream	Inlet stream
Thermal conductivity	Film[1]	Inlet stream	Average blade	Inlet stream	Inlet stream
Pressure for density	Inlet stream	Throat[2]	Mean[3]	Inlet stream	Inlet stream
Velocity	Inlet stream	Throat[2]	Exit stream	Inlet stream	Inlet stream
Characteristic length	Perimeter/π	Perimeter	Chord	Chord	Perimeter/π
Chord, in.	0.680	3.08	1.0	2.57	3.94
Number of blades in cascade	6	1	5	4	3
Chord/spacing	1.92	1.89	1.61	2.04 and 1.69	1.47
Approximate inlet static gas temperature, $^{\circ}R$	530	550	565 - 1200	535	540

[1] Film temperature, $t_f = \frac{1}{2}(t_i + t_b)$.

[2] Density multiplied by velocity, $\rho V = w/A_t$.

[3] Mean density, $\rho = \bar{p}/Rt_m$.

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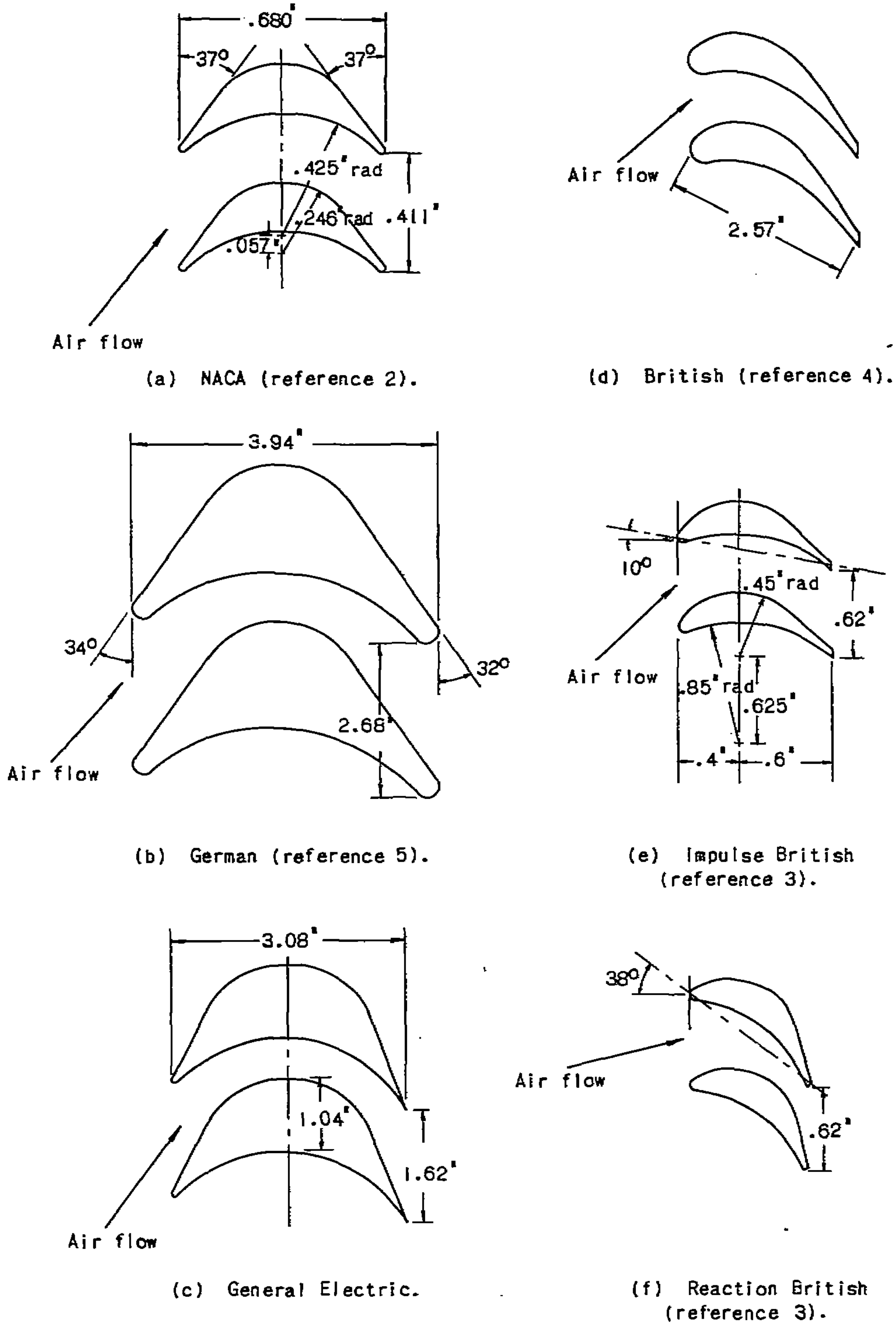
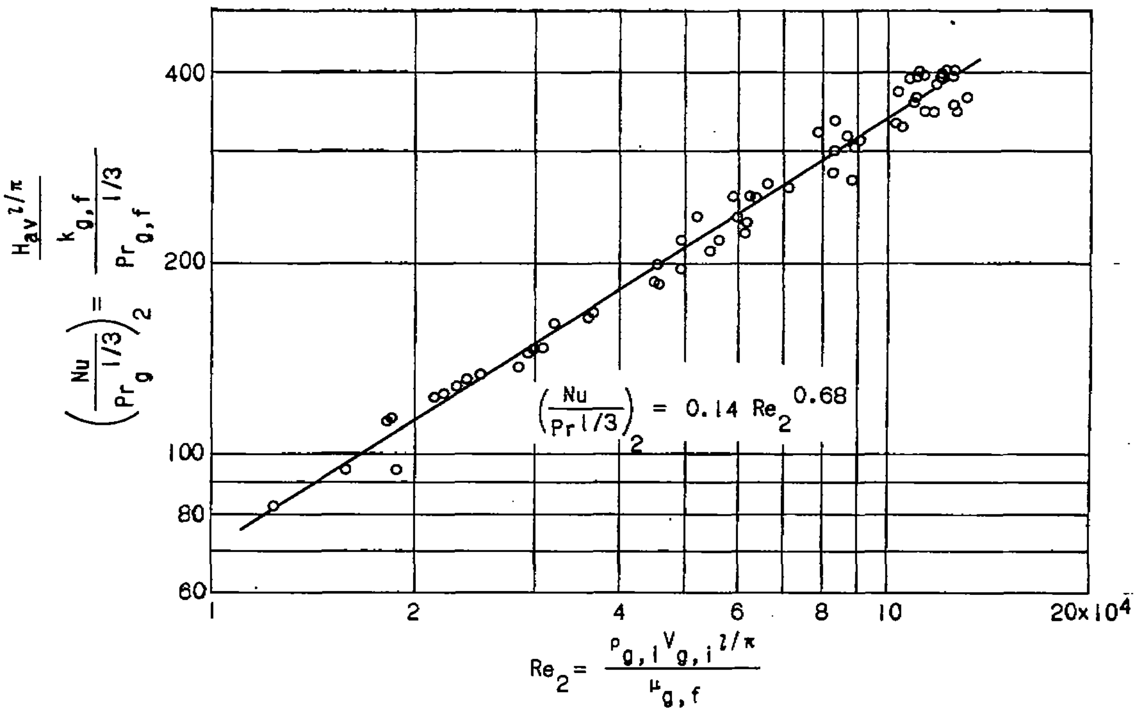
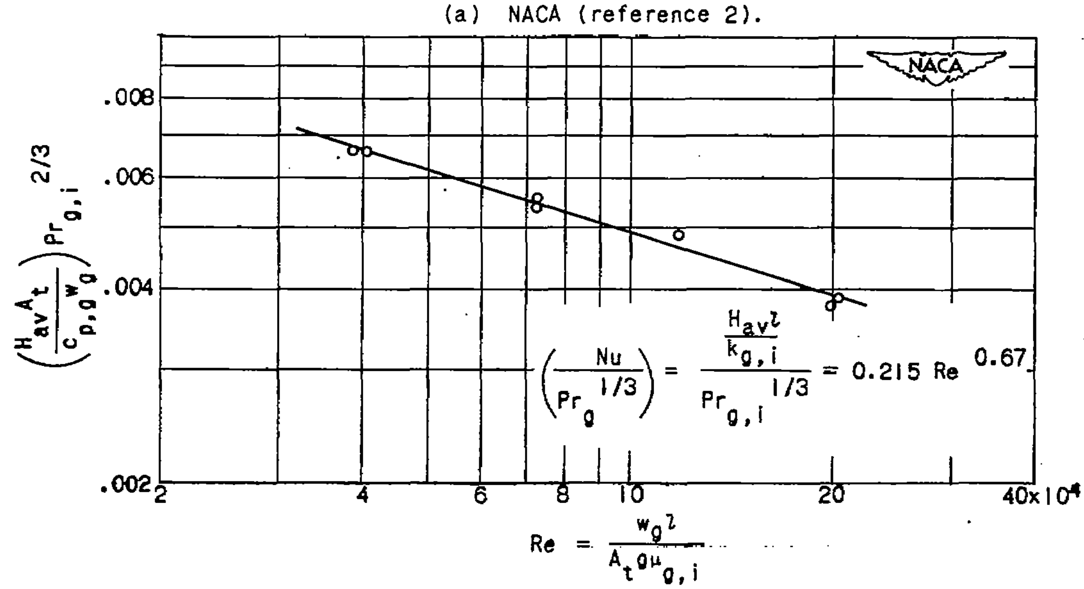


Figure 1. - Turbine-blade profiles and passage configurations.



(a) NACA (reference 2).



(b) General Electric.

Figure 2. - Experimental results of investigations.

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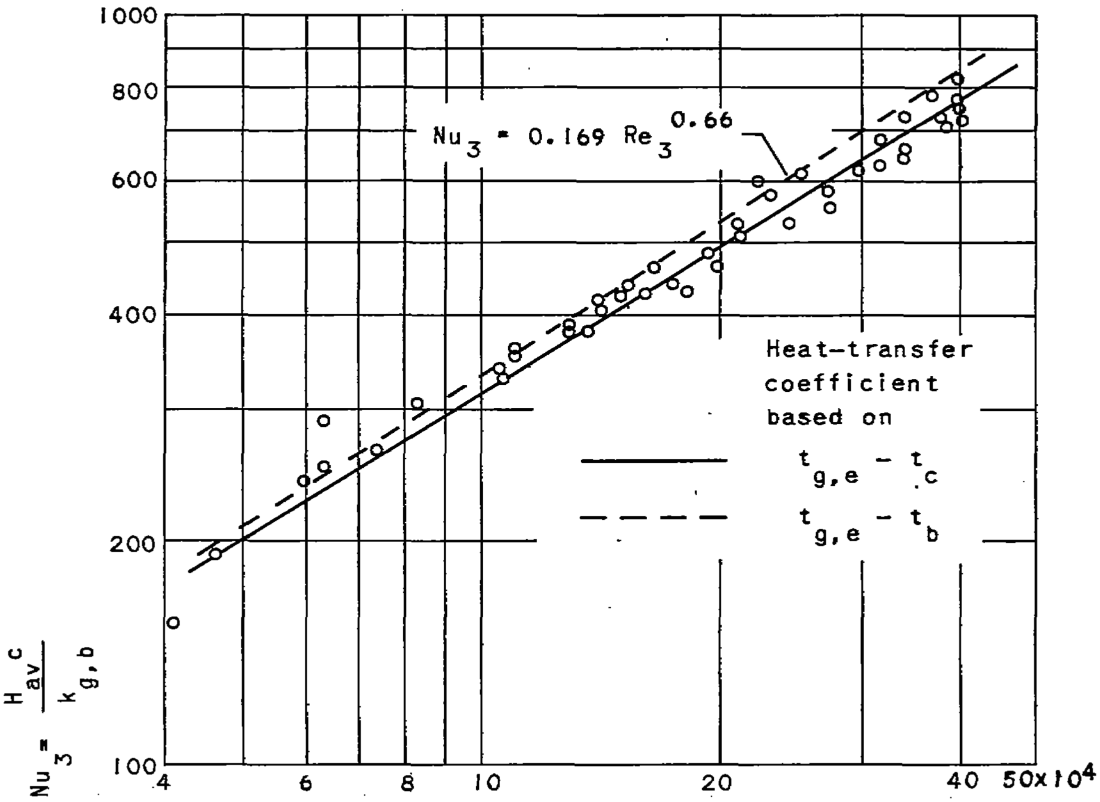
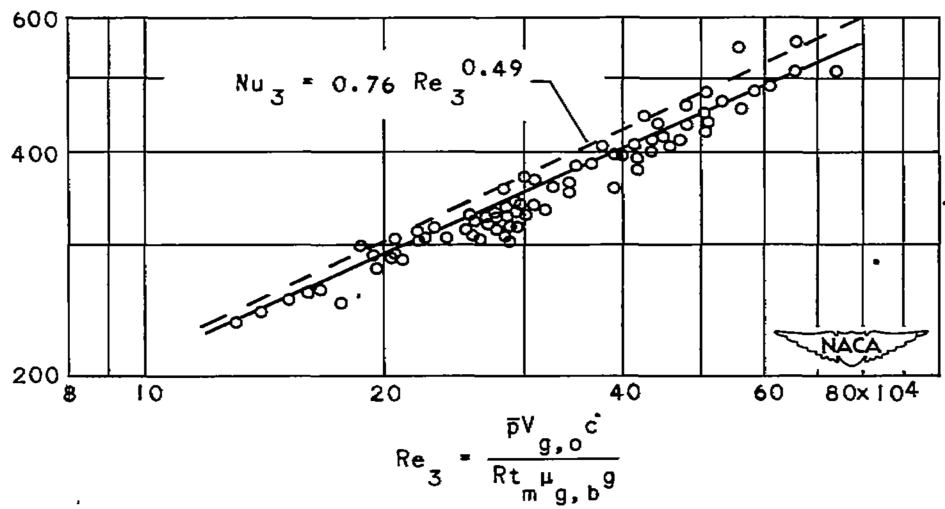
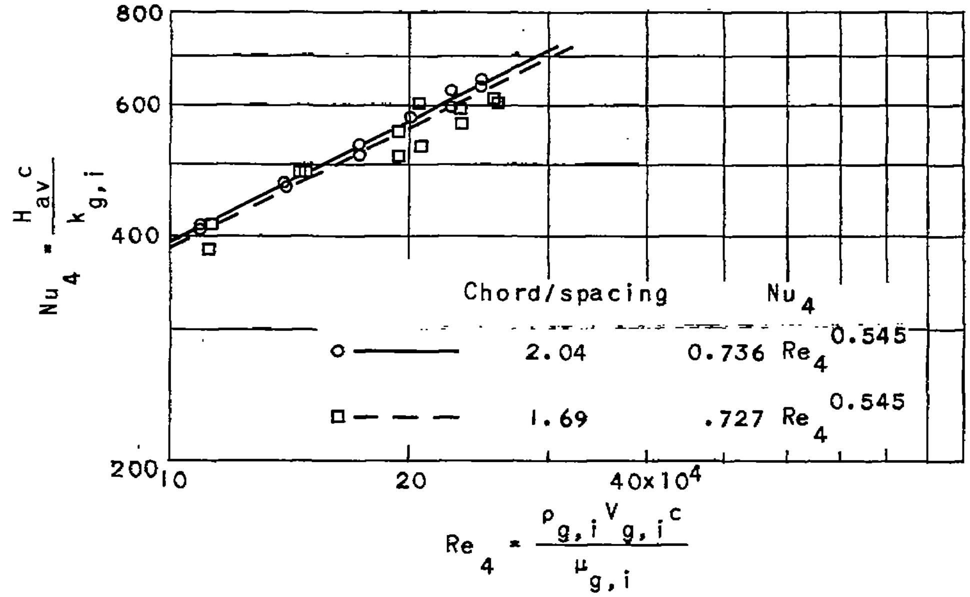


(c) British impulse blades (reference 3).

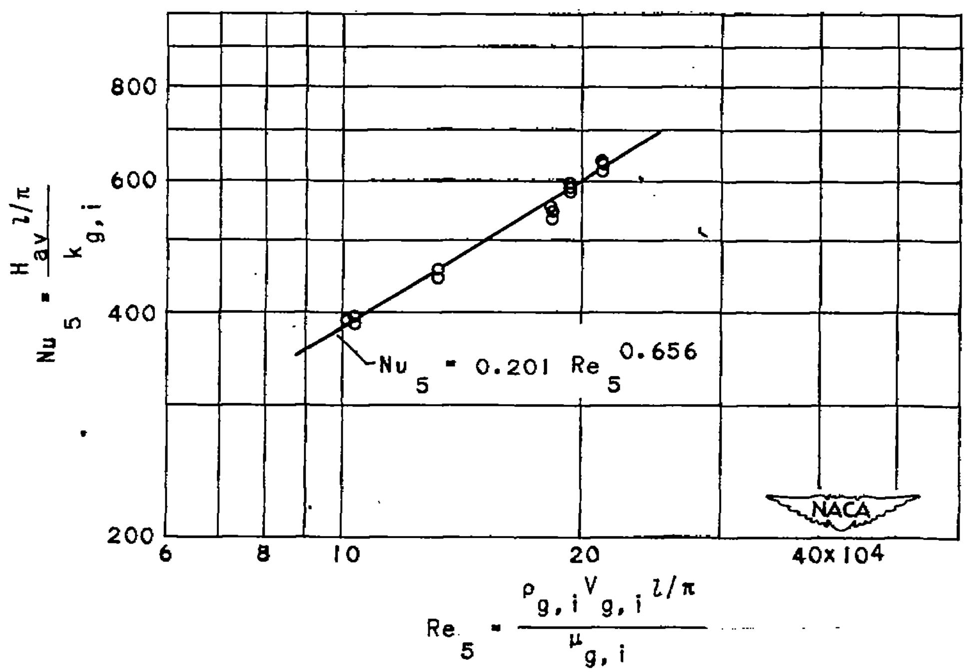


(d) British reaction blades (reference 3).

Figure 2. - Continued. Experimental results of investigations.



(e) British (reference 4).



(f) German (reference 5).

Figure 2. - Concluded. Experimental results of investigations.

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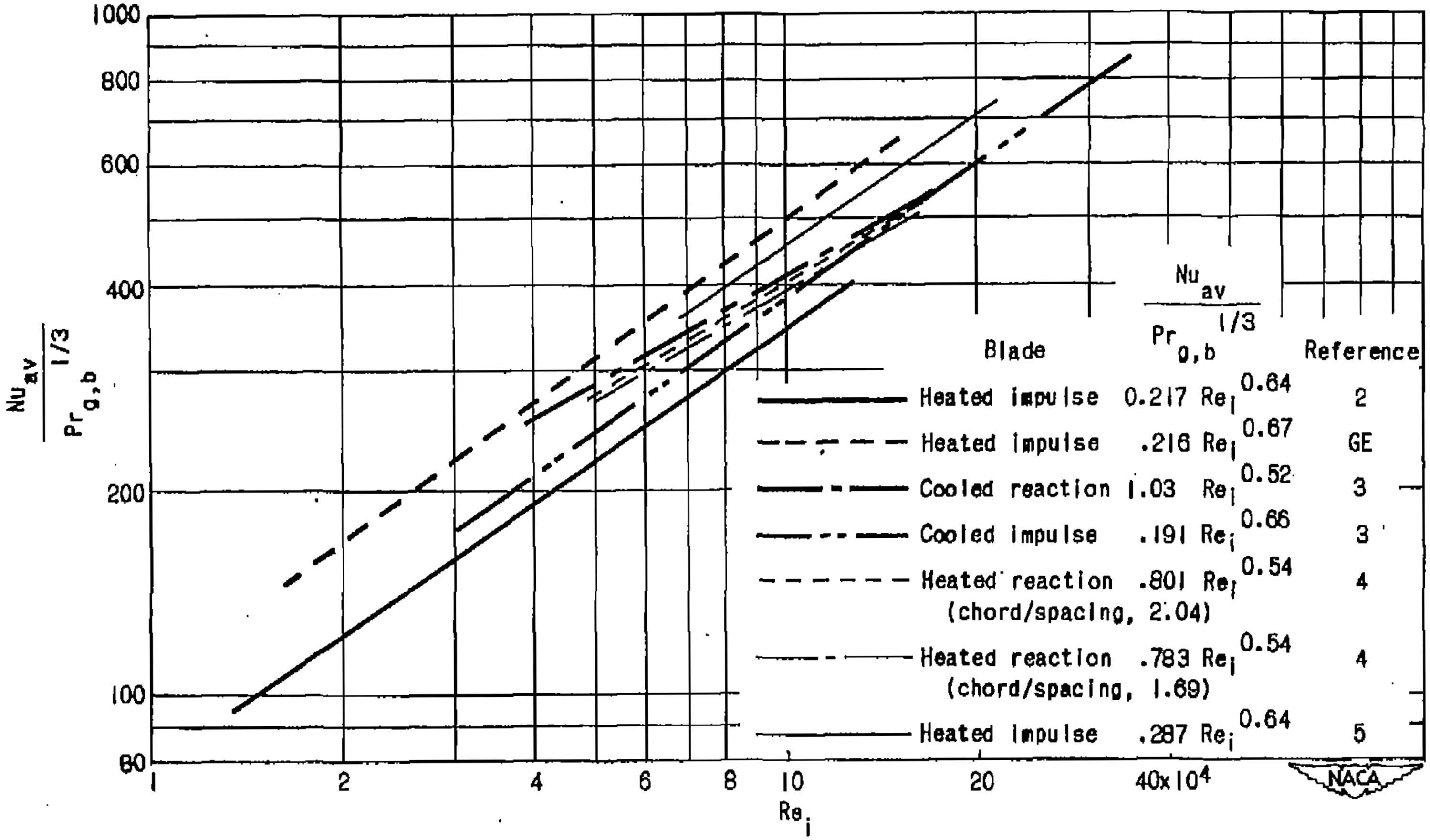


Figure 3. - Comparison of available turbine heat-transfer data computed using inlet Reynolds number.

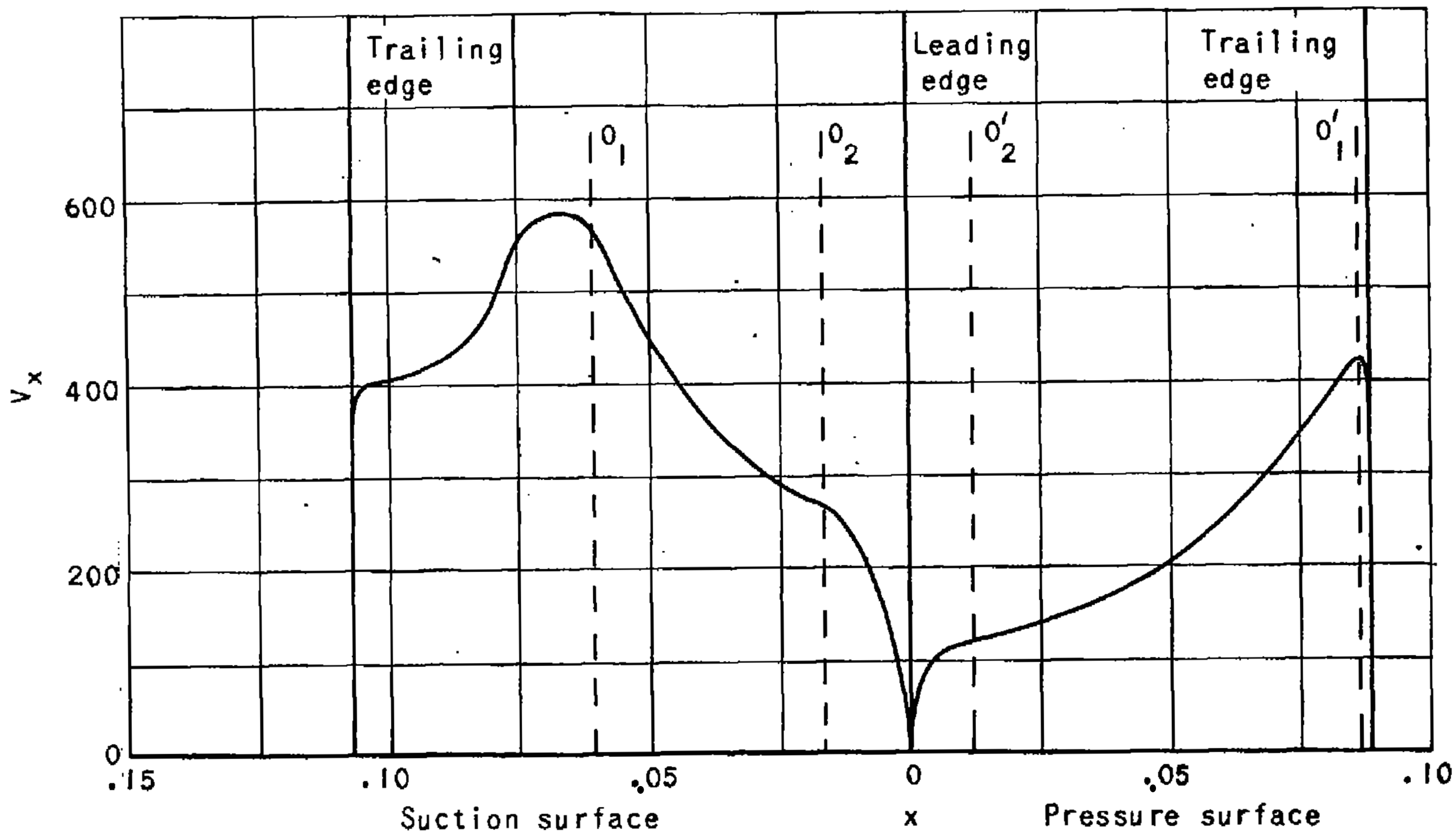


Figure 4. - Sample theoretical velocity distribution.

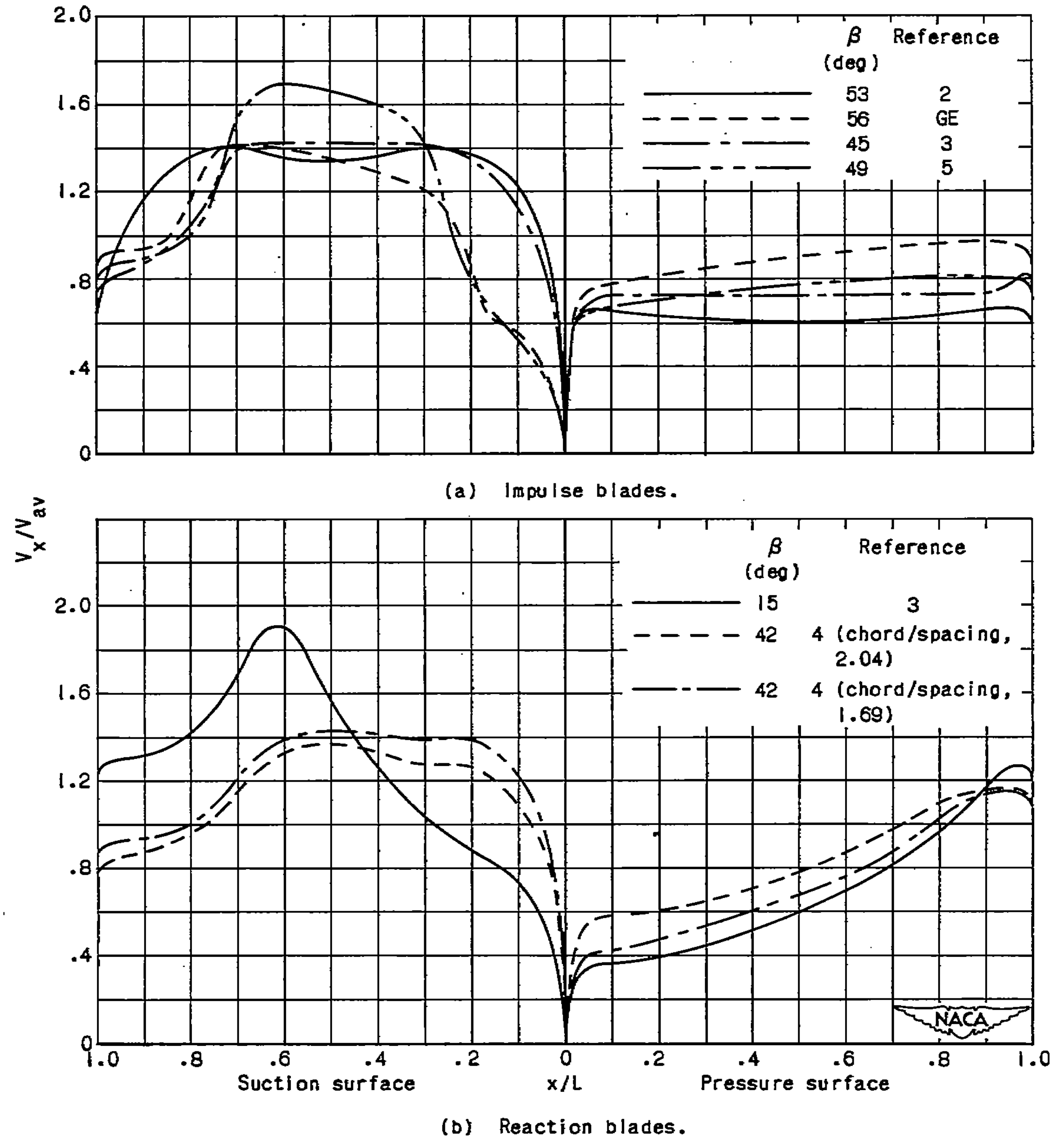
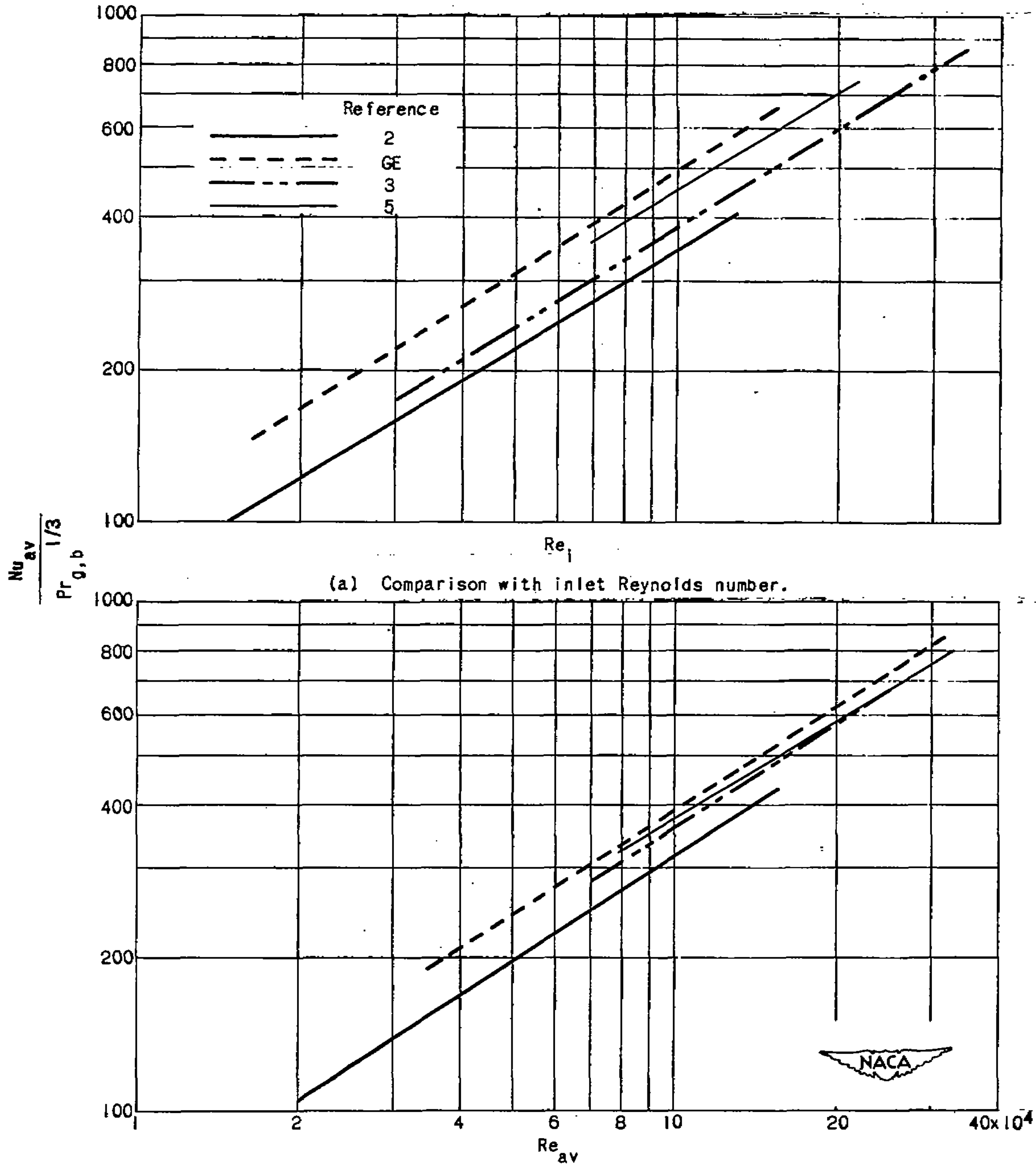


Figure 5. - Theoretical velocity distributions.



(a) Comparison with inlet Reynolds number.

(b) Comparison with average Reynolds number.

Figure 6. - Comparison of data obtained from impulse blades.

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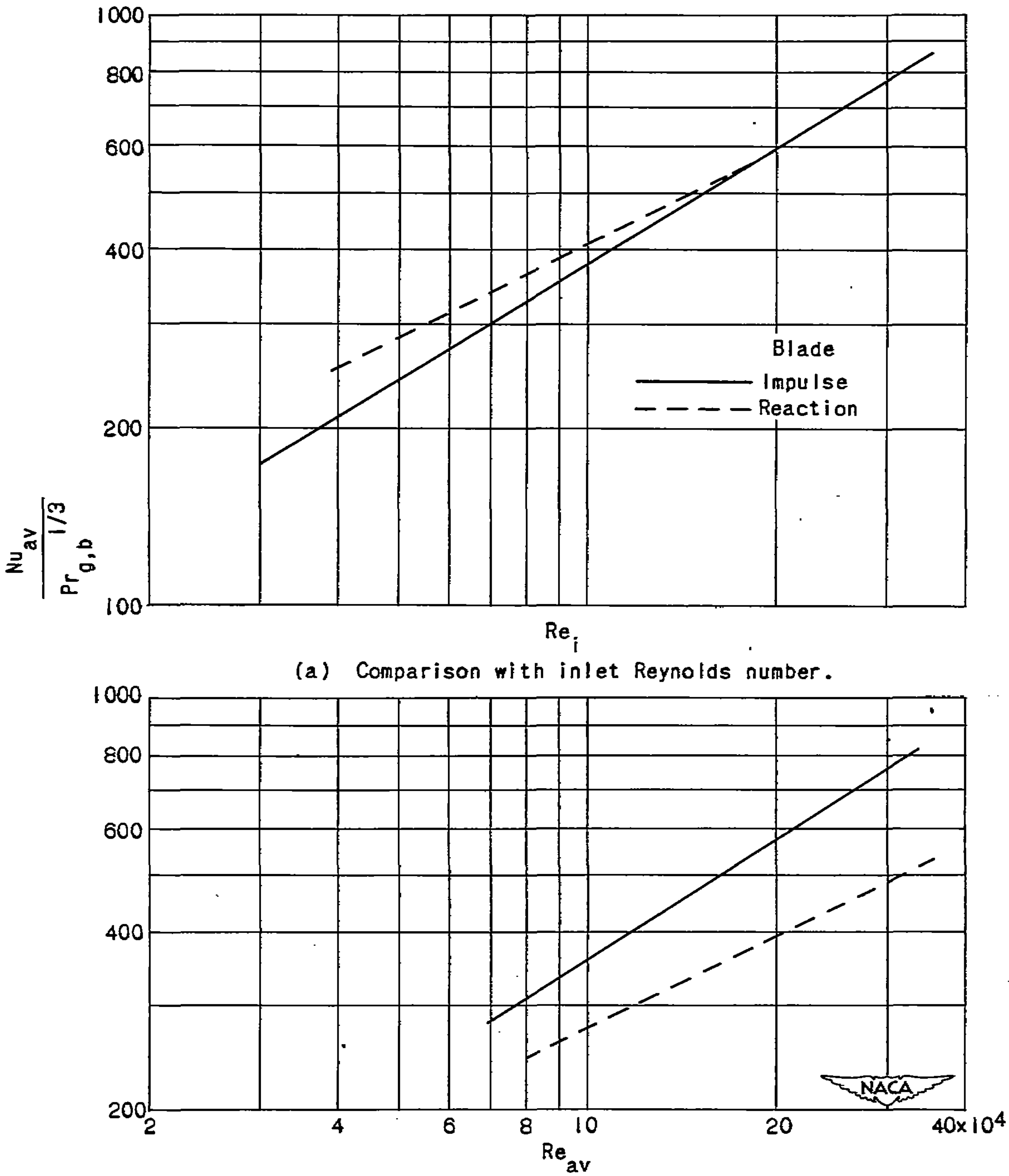

(a) Comparison with inlet Reynolds number.

(b) Comparison with average Reynolds number.

Figure 7. - Comparison of data of reference 3.

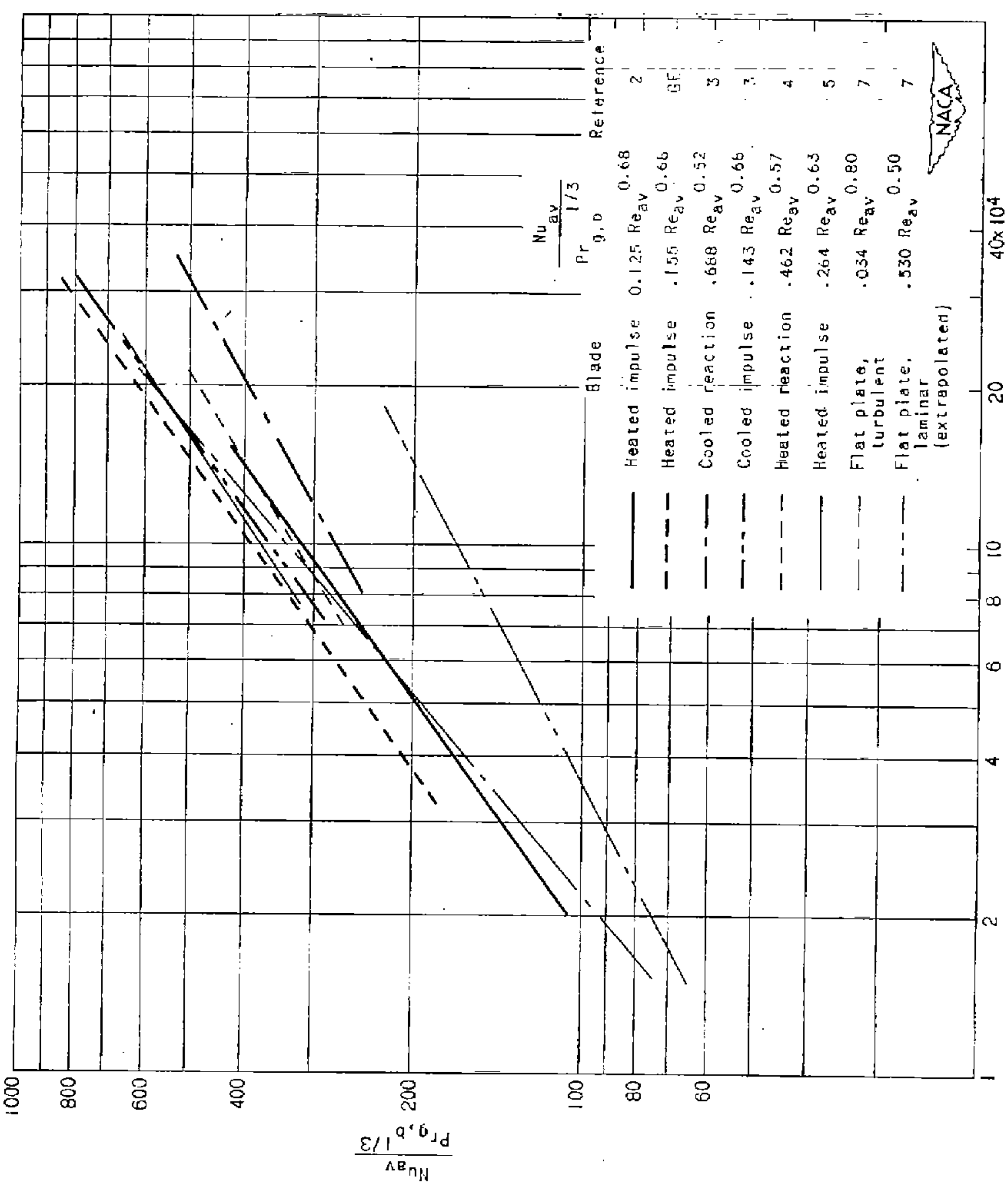

Figure 8. - Comparison of available heat-transfer results computed using average Reynolds number.

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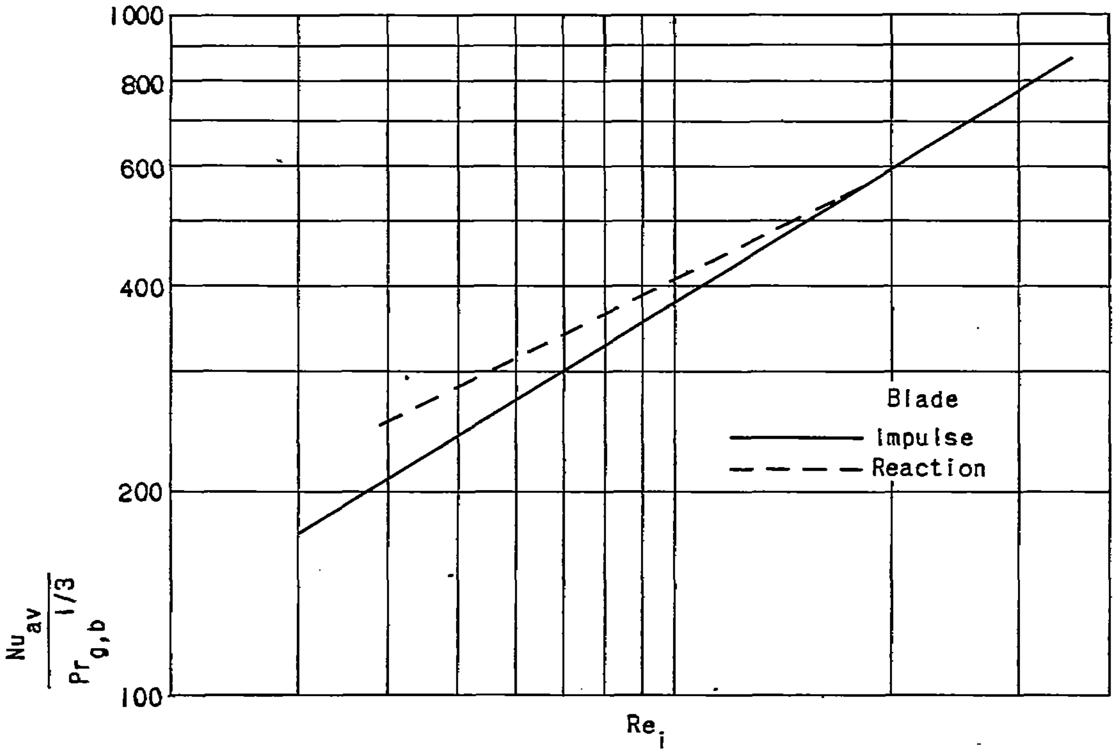
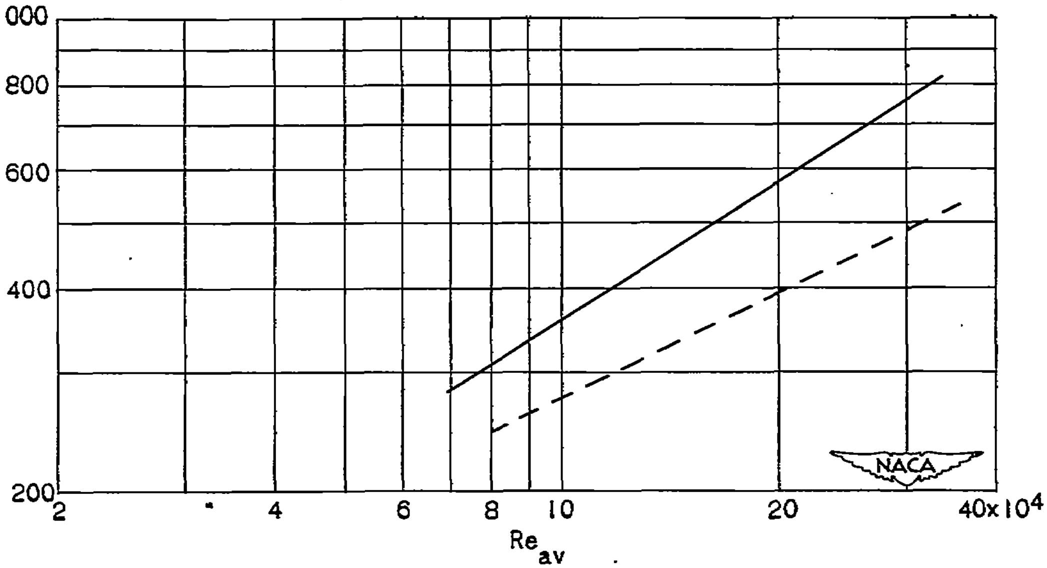


(a) Comparison with inlet Reynolds number.



(b) Comparison with average Reynolds number.

Figure 7. - Comparison of data of reference 3.

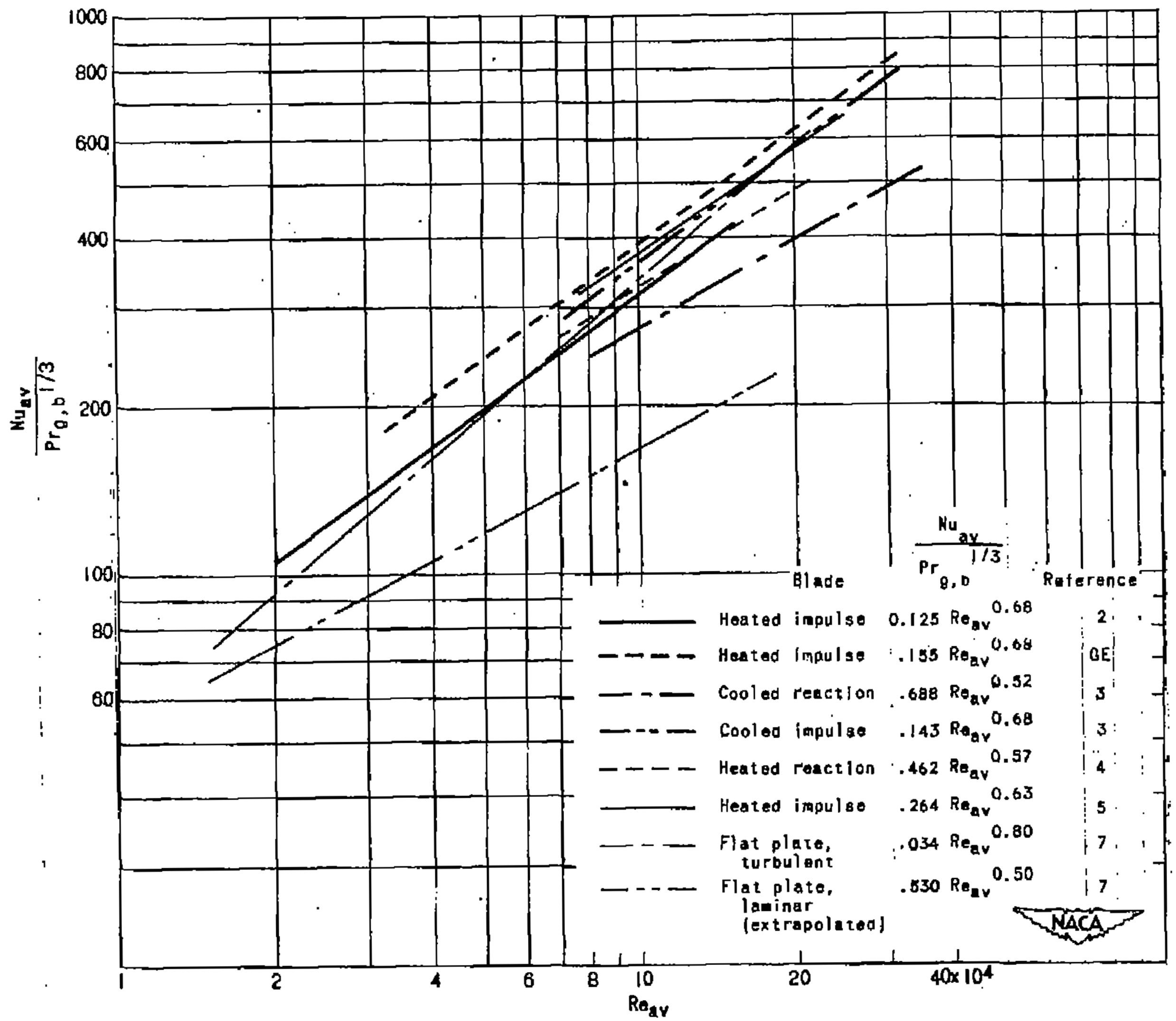


Figure 8. - Comparison of available heat-transfer results computed using average Reynolds number.

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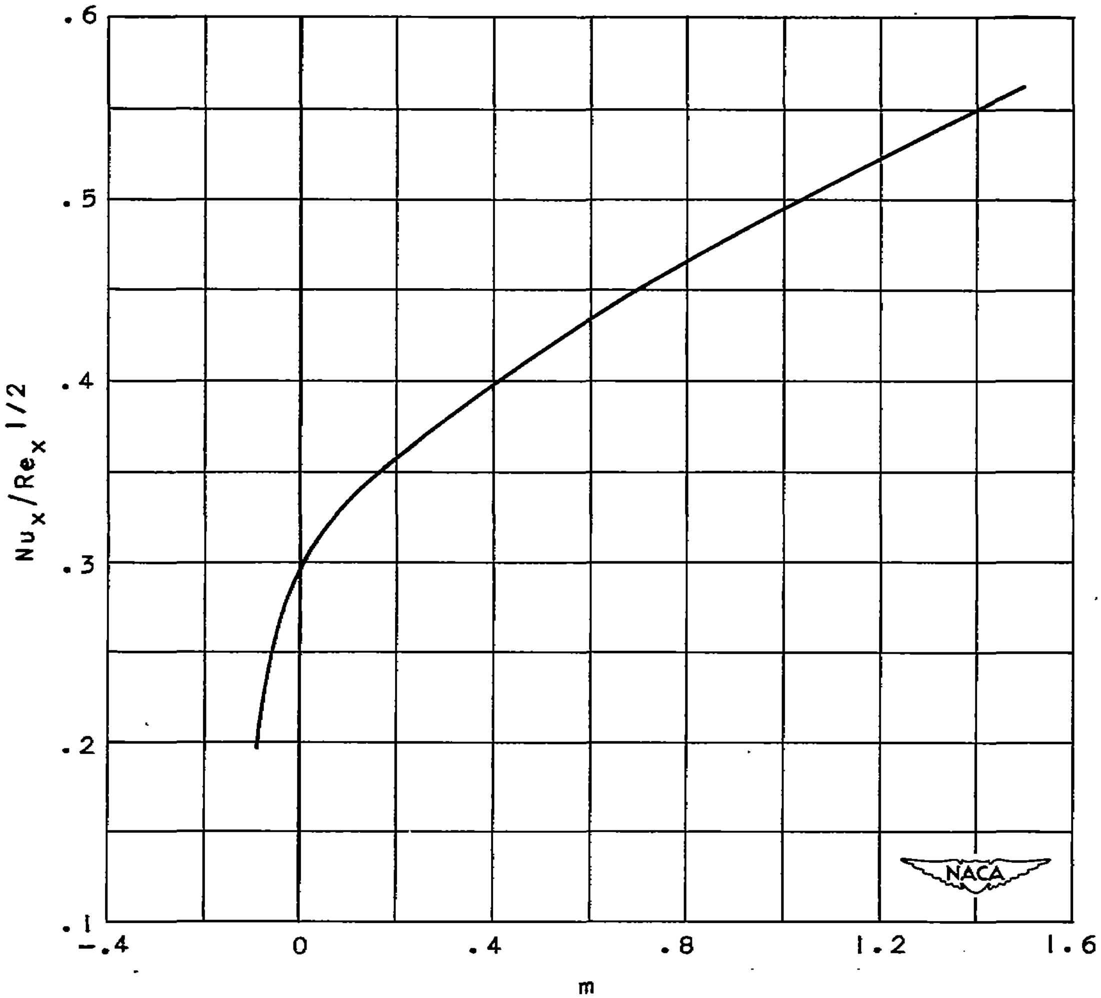

Figure 9. - Theoretical heat-transfer results of reference 8 showing pressure-gradient effects. Mach number, approximately 0; Prandtl number, 0.7; ratio of static wall temperature to static gas temperature, 1; $V_x = C_{V1}x^m$.

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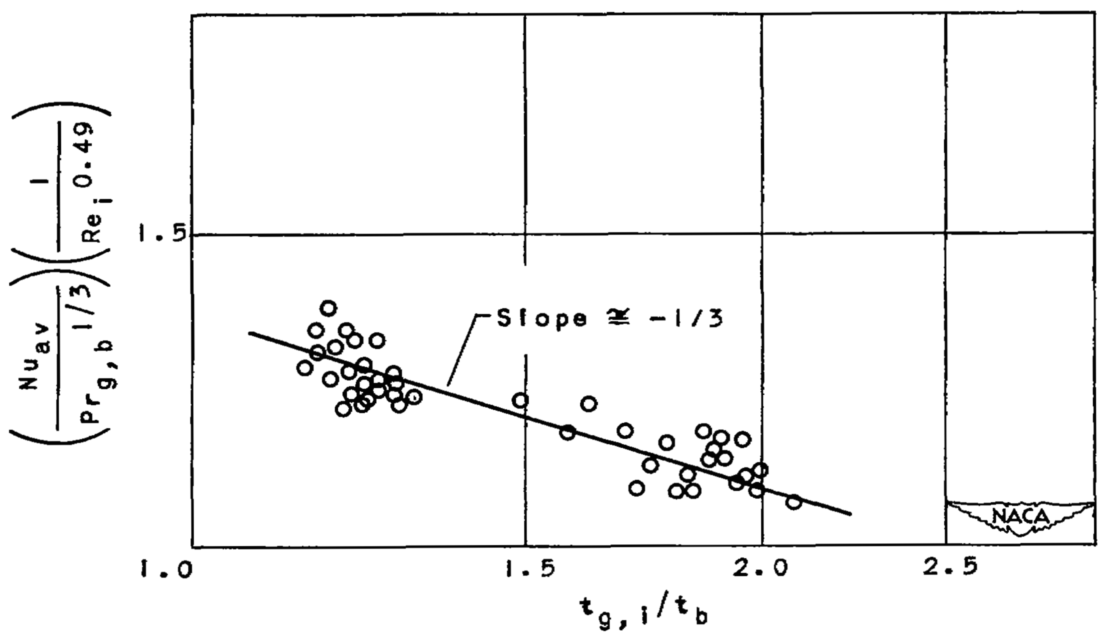

Figure 10. - Correlation of British heat-transfer data using cooled reaction turbine blades (reference 3). Gas properties based on average blade temperature. Characteristic length, perimeter divided by π.